Experimental study of an air-source heat pump for simultaneous heating and cooling - Part 2: Dynamic behaviour and two-phase thermosiphon defrosting technique

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ABSTRACT

This article presents the concepts of an air-source Heat Pump for Simultaneous heating and cooling (HPS) designed for hotels and smaller residential, commercial and office buildings in which simultaneous needs in heating and cooling are frequent. The main advantage of the HPS is to carry out simultaneously space heating and space cooling with the same energy input. Ambient air is used as a balancing source to run a heating or a cooling mode. The second advantage is that, during winter, energy recovered by the subcooling of the refrigerant is stored at first in a water tank and used subsequently as a cold source at the water evaporator to improve the average performance and to carry out defrosting of the air evaporator using a two-phase thermosiphon. Unlike conventional air-source heat pumps, defrosting is carried out without stopping the heat production. A R407C HPS prototype was built and tested. The basic concepts of the HPS are detailed in part1 of this article [1]. Its performance on defined operating conditions corresponds to the data given by the selection software of the compressor manufacturer. In the present part of this article, the operation of the high pressure control system, the transitions between heating, cooling and simultaneous modes and the defrosting sequence are analysed and validated experimentally.

Keywords: *heat pump, heating, cooling, thermosiphon, defrosting*

NOMENCLATURE

AHX	air heat exchanger
C-mode	cooling mode
E	energy (J)
h	enthalpy (kJ kg^{-1})
H-mode	heating mode
HP	high pressure (Pa)
HPS	heat pump for simultaneous heating and cooling
k	pressure drop factor (kg^2 m^{-5} s^{-2})
L	latent heat (J kg^{-1})
LP	low pressure (Pa)
m	mass (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
$\dot{Q}$	heating or cooling capacity (W)
p	pressure (Pa)
S-mode	simultaneous mode
T	temperature (°C)
t	time (s)

Subscripts:

aCd	air condenser
aEv	air evaporator
CD	condensation
cw	cold water
df	defrosting
f	frost
fus	fusion
hw	hot water
r	refrigerant
Sc	subcooler
vap	vaporisation
wCd	water condenser

wEv water evaporator

1 Context and objectives

Energy efficiency of buildings is continuously improving thanks to more and more stringent thermal regulations. Besides, comfort requirements demand more and more energy. The demand for cooling is rising to compensate internal heat gains caused by more and more household electrical equipment whilst domestic hot water (DHW) demands continue to increase. A better thermal envelope implies less energy for heating and more for cooling. Therefore thermal needs of new buildings follow the trend of more balanced energy demands between heating and cooling. ECBCS Annex 48 of the International Energy Agency deals with this issue [1]. An answer to a simultaneous energy demand in heating and cooling is the heat pump, since it has simultaneously a heating capacity at the condenser and a cooling capacity at the evaporator.

This study presents the design of a heat pump that can satisfy fluctuating needs, simultaneous or not, in heating and cooling. This heat pump with heat recovery is named HPS (Heat Pump for Simultaneous heating and cooling). It is suited to buildings such as hotels where DHW demands are high, small commercial units like groceries and petrol stations or glass-fronted north-south oriented office buildings where simultaneous needs in space heating and cooling are frequent.

Part 1 of this article [2] presents the basic concepts and the performance verification of a R407C HPS prototype. This second article validates the dynamic operation of the HPS, namely the transitions between the different modes, the high pressure control and the two-phase thermosiphon defrosting technique.

The first objective of the HPS is to produce, as often as possible, heating and cooling energies using the same electric energy input at the compressor.

The second objective of the HPS is to reduce the performance loss that characterizes air-source heat pumps during winter due to low source temperature and frosting [3,4]. Huang et al. [5] compared the most common defrosting methods: reversed-cycle defrosting and hot-gas bypass defrosting. When using the reversed-cycle defrosting technique, reversible air-to-water heat pumps record a performance loss because of a break in the heat production and a drawing of energy from the heat stock previously produced. The air and water heat exchangers have to carry out alternatively condensation at high pressure and evaporation at low pressure. When a change of mode occurs, the reversing valve switches and the previously high pressure section of the refrigeration circuit becomes the low pressure section. The opposite phenomenon occurs in the previously low pressure section. The pressure ratio changes nearly instantaneously from a high value to a very low value. This leads to high mechanical strains on the compressor that impact on its longevity. The hot-gas bypass defrosting system avoids the shocks induced by the switch of the reversing valve and the heat loss in the hot water tank that would occur in a standard reversible heat pump. However a performance loss remains because of the bypass of hot vapour from the compressor discharge. Liang et al. [6] detailed another defrosting method: hot gas from the discharge line is bypassed, throttled and injected into the evaporator to provide sensible heat as a defrosting energy. Attention is especially paid on the fuzzy logic regulation strategy. Electric heating defrosting can also be used indirectly by heating the air flowing throughout the outdoor coil [7]. Another technique described by Liu et al. [8] consists in re-circulating exhaust air in the air evaporator in order to reduce the frost growth and to delay the defrosting sequence. All conventional defrosting methods consume energy. The HPS proposes a special circuit design that creates a two-phase thermosiphon used for defrosting. In refrigeration plants, thermosiphons are usually cancelled out by non-return valves because they represent risks of liquid refrigerant trapping in the circuit. The HPS takes advantage of this phenomenon to carry out defrosting. Thermosiphons are known to be an efficient means of heat transfer [9,10]. Dobson [11] showed that thermosiphon effects were difficult to model. He also observed that the flows were oscillatory.

The aim of this phenomenological study is to verify that the high pressure control system is efficient, that transitions between modes are sufficiently smooth and to validate the innovative defrosting technique using a two-phase thermosiphon.

2 Further in the HPS concepts

The HPS prototype scheme is shown in Fig. 1. The different components of the circuit and the operating modes (heating, cooling and simultaneous modes) are deeply described in the first part of this article [2]. The heating mode uses the air evaporator and the water condenser (Evr2 and Evr3 are open). The cooling mode uses the water evaporator and the air condenser (Evr1 and Evr4 are open). The simultaneous mode uses the water evaporator and the water condenser (Evr1 and Evr3 are open). The present paragraph explains the dynamic behaviour of the HPS depending on the needs in heating and cooling. The operation is based on the alternation of the heating, cooling and simultaneous modes with or without the two-phase thermosiphon defrosting technique. Switching from one mode to another is simply carried out by simultaneously shutting and opening the appropriate electronic valves without stopping the compressor. A high pressure control system, described hereafter, is necessary to ensure the proper operation of the HPS.

2.1 Summer sequence

During summer, the sequence alternates between cooling and simultaneous modes. A cooling demand is detected by a cold water temperature higher than the set temperature. A summer sequence is then launched, beginning by the simultaneous mode in case of possible demand in heating. When the heating demand is satisfied, the HPS switches to the cooling mode and the air condenser is used instead of the water condenser. The HPS stops when the cooling demand is satisfied or switches back to the simultaneous mode if a heating demand is detected by the measurement of a low hot water temperature.

2.2 Classic winter sequence

During winter, the sequence alternates between heating and simultaneous modes. The cold water tank is used as a short-time heat storage. If a heating demand is detected, the sequence begins by the heating mode engaging the water condenser, the air evaporator and the subcooler. While producing hot water using the water condenser, the cold water tank is heated, usually from 5 to 15°C, by the refrigerant subcooling energy. When the temperature of 15°C is reached, the HPS regulation switches to the simultaneous mode and uses the energy previously stored in the cold water tank as a cold source for the water evaporator. The cold water tank temperature decreases from 15 to 5°C.

In the simultaneous mode, the evaporating temperature is higher than in the heating mode from the moment that ambient air is colder than the cold water tank. Therefore, using the simultaneous mode for a certain time during the winter sequence enables to produce hot water continuously with improved average system performance compared to standard air-source heat pumps. Besides, in the simultaneous mode, the air evaporator is not used for evaporation and can be defrosted using a two-phase thermosiphon detailed hereunder.

2.3 Winter sequence with defrosting

In the heating mode, under cold outside air temperatures, the fins of the air evaporator get frosted. Before frost thickness becomes critical, the cold tank temperature has raised 15°C and the simultaneous mode has been engaged. In this mode, the air coil is automatically defrosted by a two-phase thermosiphon formed between the two evaporators. A supplementary flow of vapour comes out of the water evaporator and migrates towards the air evaporator where the temperature, and thus the pressure, is lower. At the outlet of the water evaporator, the refrigerant vapour temperature is between 10°C and 5°C depending on the cold water temperature. The refrigerant exchanges latent heat with the frosted fins, condenses and flows back to the water evaporator or to the suction accumulator by gravity.

The major advantage of this sequence is to carry out defrosting without stopping heat production. Frost thickness can thus be minimized and mean convection heat transfer coefficients at the evaporator can be maximized. The average heat pump efficiency under frosting conditions is improved compared to the performance of standard air-source heat pumps that use hot gas or reversed cycle defrosting methods.

2.4 High pressure control

As pressures and temperatures are linked during condensation, high pressure control can ensure that condensation occurs in the appropriate heat exchanger. Moreover it is able to control the condensation temperature and thus the heating capacity. A special liquid receiver is placed on the liquid line. It is connected to the compressor discharge line and the inlet of the air evaporator by copper tubes of smaller diameter on which normally closed electronic on-off valves are placed (EvrHP and EvrLP on Fig. 1).

If the chosen mode is the heating mode, condensation occurs in the water condenser. The controller calculates the set point for high pressure, function of the hot water inlet temperature at the water condenser (Eq. (1)) and a temperature discrepancy characterized experimentally. The equation is a curve fit of data extracted from tables published by the International Institute of Refrigeration [12]. The discrepancy ΔT_{CD} takes into account the pinch difference between hot water inlet temperature and bubble temperature of refrigerant, a slight subcooling because of a possible heterogeneity of phase in the tube at the location where the temperature sensor is placed and a security in relation to the control deadband. For condensation at the air coil in the cooling mode, the set point for high pressure is a function of the air inlet temperature and another temperature discrepancy proper to the heat exchanger.

$$HP_{Set} = 0.000011 \cdot \left(T_{hw-in} + \Delta T_{CD}\right)^3 + 0.0023 \cdot \left(T_{hw-in} + \Delta T_{CD}\right)^2 + 0.1833 \cdot \left(T_{hw-in} + \Delta T_{CD}\right) + 5.6738 \tag{1}$$

The objective of the high pressure control is to carry out a complete condensation in the water condenser. Without this system the thermostatic expansion valve would have “naturally” controlled the system so that condensation would finish in the subcooler. Some latent heat from condensation would have not been directly transferred to the hot water.

The high pressure control system indirectly controls the volume of liquid in the receiver. The volume of liquid in the different condensers depends on the mode. If the high pressure is below the set point, the electronic valve EvrHP is opened. The receiver is filled up with gas coming from the compressor discharge line at a pressure higher than the pressure in the receiver until the set pressure is reached. The gas entering the receiver drives the liquid towards the evaporator. The non-return valve closes because pressure becomes higher at the outlet than at the inlet. The subcooling heat exchanger and the bottom part of the water condenser are filled up with more liquid until the appropriate level of liquid is reached. If however the chosen mode is the cooling mode, the condenser becomes the air heat exchanger. The set point for pressure is then the lowest possible. The electronic valve EvrBP is opened. The pressure is reduced by driving the vapour out of the top part of the receiver towards the inlet of the air evaporator. The refrigerant in a liquid phase is sucked out of the water condenser and the subcooler and enters the receiver.

The proper operation of the control system depends upon a special liquid receiver being designed quite high and quite narrow with the main objective to enhance temperature stratification and to limit as far as possible thermal transfer between the gas and the liquid. When the gas is injected, part of it condenses. When gas is rejected to the low pressure, part of the liquid evaporates. These phenomena reduce the liquid variation induced by a pressure variation in the receiver. Although it delays the response in terms of high pressure, it ends by stabilizing the control system. The receiver is also thermally insulated to reduce the heat transfer towards the ambience.

3 Experimental study

3.1 Experimental setup

The experimental setup is described in the first part of this article [2].

3.2 Dynamic behaviour

The operating sequences were tested, verifying the operation of the high pressure control method and the transitions between modes. Fig. 2 presents the evolution of temperatures of water and air sources during a 10-minute sequence engaging first the heating mode, followed by the simultaneous mode. The acceptable operation of the high pressure control system is also shown in this figure by the measurement of hot gas temperature at the top of the liquid receiver. The set pressure increases following Eq. (1) depending on the hot water temperature at the entrance of the condenser. The hot gas temperature increases when the electronic valve EvrHP opens and lets hot gas enter the receiver and decreases with thermal exchange with the environment. Hot gas injection frequency is higher during the heating mode than during the simultaneous mode because of the position of the liquid receiver in the system: the receiver was placed in the air flow. During the heating mode the air evaporator and the fan are in function. Thermal exchange between the liquid receiver and the environment is enhanced during this mode. During the simultaneous mode the air circulation is stopped. Less thermal losses are thus observed. Fig. 2 highlights that the prototype has to be modified to reduce thermal losses of the liquid receiver and that the injection frequency is too high (between 25 and 40 seconds) to ensure a satisfactory longevity to the electronic valve EvrHP.

When the electronic valve EvrLP opens and drives out gas from the receiver towards the entrance of the air evaporator, the high pressure decreases. Fig. 2 does not show this operation that is actually used during transitions between modes having different hot sources.

In order to present each transition fully, figs. 3 and 4 show the temperatures of refrigerant and sources during transitions between heating and simultaneous modes and between cooling and simultaneous modes. Globally, temperature variations are smooth for both water and refrigerant at the inlets and the outlets of the water condenser and the water evaporator. Stronger temperature variations appear on the air source and for refrigerant at the air condenser. The latter temperature variation is due to condensation of refrigerant in the air condenser even when it is unused in the simultaneous mode. Temperature and thus pressure decreases while refrigerant is condensing. In this situation, a mechanical shock can occur in the circuit if the electronic valve Evr4 opens. The strategy adopted to avoid this risk is to use the control system to decrease the high pressure to the level of the air heat exchanger side before opening Evr4.

The dynamic behaviour of the HPS is validated because the objectives of an efficient high pressure control system and smooth transitions are satisfied. Oil return has also been checked using the oil light of the compressor: no low level was noticed.

3.3 Defrosting technique

3.3.1 Two-phase thermosiphon observation

The two-phase thermosiphon circulates in tubes that have large diameters (represented in green in Figs. 1 and 5). Vapour migrates from the water evaporator towards the air evaporators. Condensed refrigerant returns back by gravity to the water evaporator and the suction accumulator.

Thermographic pictures (Fig. 6) have been taken before and during the defrosting phase at the position of the white square in Fig. 5. Before the defrosting sequence, temperature is homogenous. The tube contains vapour flowing from the air evaporator towards the compressor. When the two-phase thermosiphon defrosting sequence is launched, the thermostatic expansion valve TEV2 does not supply refrigerant to the air evaporator anymore because the electronic valve Evr2 is closed. In Fig. 6 it can be seen that heterogeneity of temperature appears in the tube. The hotter vapour phase flows up the tube in the top part of the section and the colder liquid phase flows down in the bottom part.

3.3.2 Defrosting technique validation

The validation is based on the observation of the frost layer disappearance during a defrosting sequence using the two-phase thermosiphon. The defrosting time is compared to a frost layer disappearance time by thermal exchange with the ambient air surrounding the test bench. In this second case compressor and fans are stopped.

Frosting: The first phase consists in running the heat pump in a heating mode under frosting conditions. Ambient air temperature is controlled around 0°C. Frost progressively appears on the LP sections of the air coils of the prototype. The frost layer thickness increases over a period of 30 minutes.

Defrosting (Fig. 7): After running the heating mode during 30 minutes, the frost layer is assumed to be sufficient to run the defrosting sequence. Fig. 7 shows heterogeneity of the frost layer due to an incorrect tubing of the three-fluid AHX. The electronic valve Evr2 is switched off and the electronic valve Evr1 is switched on. The water evaporator is used instead of the air evaporator. The simultaneous mode is engaged and the two-phase thermosiphon defrosting starts. The frost layer disappears within 2 minutes. As a comparison, after a 30-minute frosting phase, the prototype was stopped. The defrosting energy was then brought by thermal convective exchange with the ambient air surrounding the test bench. The time of frost disappearance reached 20 minutes. This comparison confirms that the thermosiphon defrosting technique works very efficiently.

3.3.3 Two-phase thermosiphon characterization

The frost mass is evaluated by the mass of water collected after defrosting. The defrosting energy E_{df} is calculated by Eq. (4) using the frost mass m_f and the specific latent heat of fusion L_{fus}.

$$E_{df} = m_f \cdot L_{fus} \tag{4}$$

The average defrosting capacity is calculated by Eq. (5) using the defrosting energy E_{df} and the defrosting time t_{df}.

$$\overline{\dot{Q}_{df}} = \frac{E_{df}}{t_{df}} \tag{5}$$

The mean refrigerant mass flow rate is given by Eq. (6) using the enthalpy difference between saturated vapour and saturated liquid at 0°C.

$$\overline{\dot{m}_r} = \frac{\overline{\dot{Q}_{df}}}{\Delta h_{vap}(0°C)} \tag{6}$$

During the defrosting sequence, refrigerant temperatures have been monitored at the outlets of the air evaporators T_{aEv} and the water evaporator T_{wEv} (Fig. 1) Corresponding pressures are approximated using the vapour saturation P-T curve. The vapour mass flow rate at time i depends on the pressure difference using Eq. (7).

$$\dot{m}_r{}^i = \sqrt{\frac{(p_{wEv} - p_{aEv})^i}{k^i}} \tag{7}$$

This equation depends on the pressure drop factor k^i. This factor is assumed to be constant during the defrosting sequence because the section of the tube is negligibly reduced by the flow of liquid that returns back from the air evaporator by gravity. Moreover thermal exchange between the two phases, flowing up (vapour phase) and down (liquid phase), has not been studied. Following these assumptions, the refrigerant mass flow rate can be estimated using Eq. (8). Defrosting capacity is given by Eq. (9). These two equations bring into play the average pressure difference between the outlets of the water evaporator and the air evaporator.

$$\dot{m}_r{}^i = \overline{\dot{m}_r} \cdot \sqrt{\frac{(p_{wEv} - p_{aEv})^i}{\overline{(p_{wEv} - p_{aEv})}}} \quad (8)$$

$$\dot{Q}_{df}{}^i = \overline{\dot{Q}_{df}} \cdot \sqrt{\frac{(p_{wEv} - p_{aEv})^i}{\overline{(p_{wEv} - p_{aEv})}}} \quad (9)$$

Table 1 presents the results for frost mass, defrosting energy and mean values of mass flow rate and defrosting, cooling and heating capacities. Mean cooling and heating capacities are measured on water. These values show that there is no break in the heat production and that the heat source has switched from ambient air to cold water. Fig. 8 shows the evolutions of the refrigerant mass flow rate and the defrosting capacity during the 2-minute thermosiphon defrosting sequence. The defrosting capacity curve decreases from 2.32 to 0.26 kW. The mass flow rate curve decreases from 6.28 to 2.08 g.s^{-1}. The important variations between 50 s and 100 s are due to the internal pressure control, activated after switching from the heating mode to the simultaneous mode. After 120 s, the temperature difference between the refrigerant temperatures at the outlets of air and water evaporators (respectively T_{aEv} and T_{wEv} in Fig. 1) is around 1°C, which is within the limit of uncertainty of the metrology. The operating conditions at the beginning of the defrosting sequence are calculated using the software of the compressor manufacturer. They are reported in table 2. According to these values, the supplementary refrigerant mass flow rate to be generated at the water evaporator for the thermosiphon represents 6% of the flow rate at the compressor. Therefore the water evaporator has to be at least oversized in this proportion. The initial defrosting capacity corresponds to more than 12% of the nominal cooling capacity.

4 Conclusions

This article is the second part of an experimental study of an air-source heat pump for simultaneous heating and cooling (HPS). The first part detailed the basic concepts and the performance validation of the HPS. The present article goes further and deals with the operating sequences alternating between heating, cooling and simultaneous modes. Apart from producing simultaneously hot and cold water, this machine proposes an answer to reduce the performance loss of air-to-water heat pumps under low ambient temperatures by the use of operating sequences alternating between an air evaporator and a water evaporator. Moreover, while standard air-source reversible heat pumps are penalized by defrosting (break in the heat production, decrease in COP), the HPS carries out defrosting using a two-phase thermosiphon with increased heat pump performance and without stopping the heat production. Thanks to the special management of produced energies, simulations showed an annual performance increase of 16.6% when comparing the HPS to a standard reversible heat pump [13].

The phenomenology of the HPS operation has first been studied on two points: high pressure control and transitions between modes. The high pressure control is operational but needs more development. Transitions between modes are smooth, thus acceptable.

Then the two-phase thermosiphon has been observed by thermographic pictures of a section of the tube between the two evaporators and validated by a favourable defrosting time in comparison with a situation in which the heat pump was stopped. The characterization of the thermosiphon showed a phenomenon of high capacity. Nevertheless these encouraging results still need to be comforted by in situ tests. Further research has to be conducted on the two-thermosiphon defrosting technique. More metrology has to be implemented to improve the characterization. The validation has to be conducted under higher levels of frosting using other experimental means such as a climatic room. Finally, R407C is not a good choice for the HPS because the two-phase thermosiphon will provoke a separation of the different components of the refrigerant mixture [14]. Eventually, the performance and even the proper operation of the HPS can be altered.

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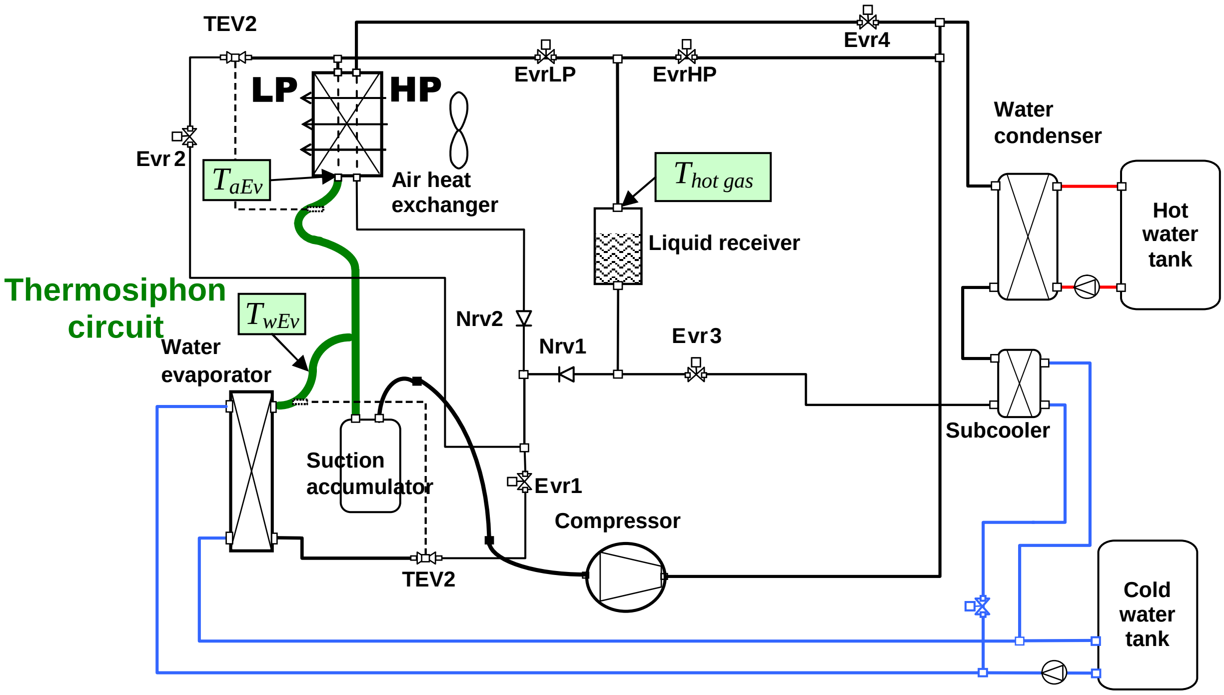


Fig. 1. HPS circuit

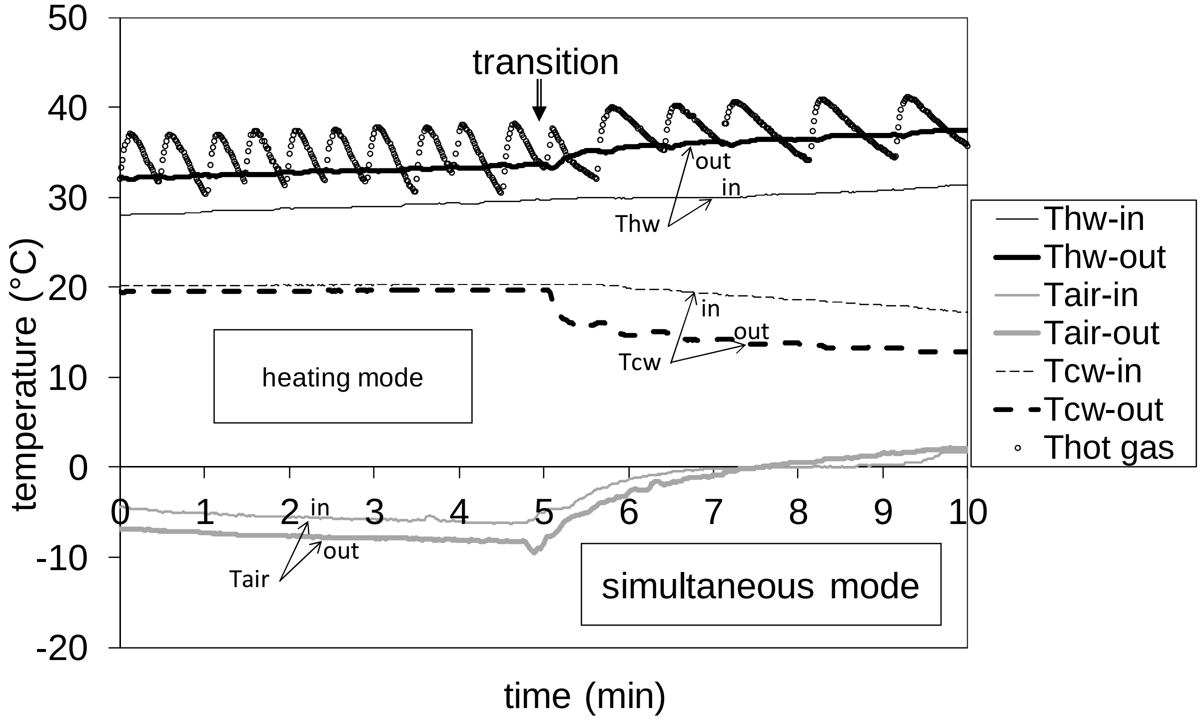


Fig. 2. Source temperatures and hot gas temperature at the receiver during a transition between the heating mode and the simultaneous mode

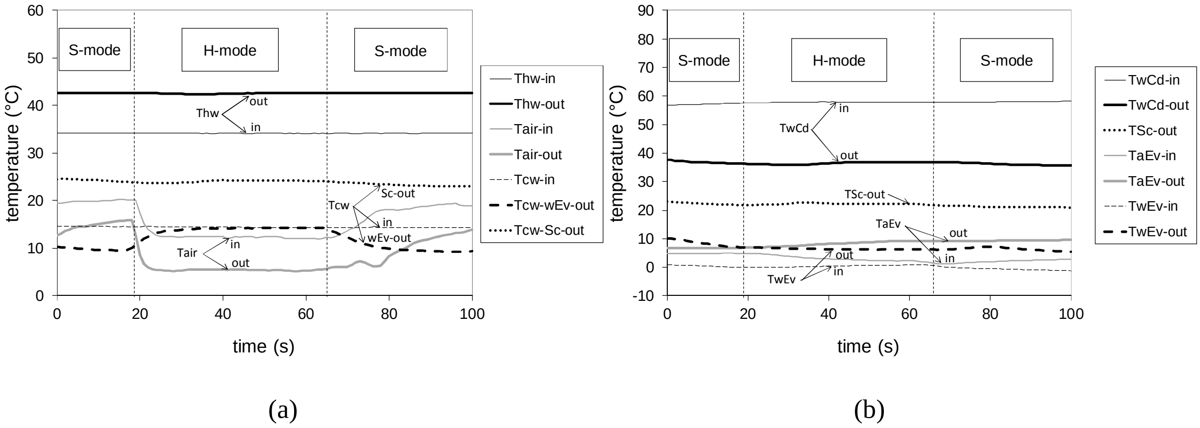
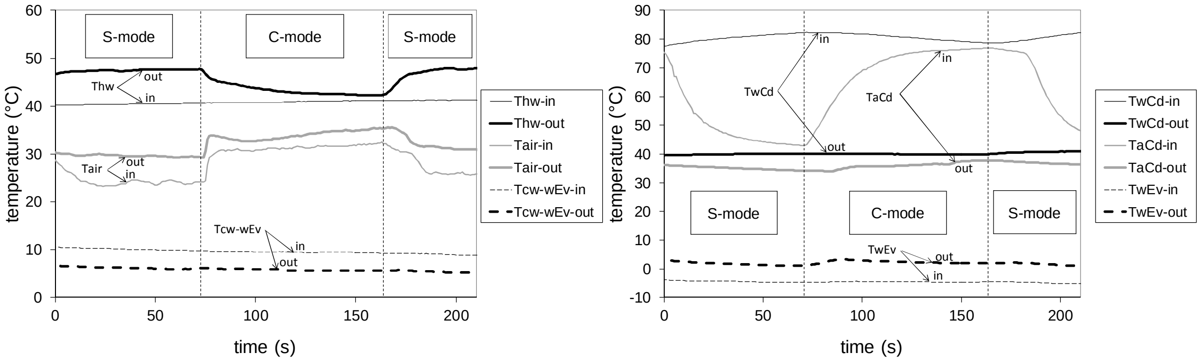


Fig. 3. Measured temperatures of sources (a) and refrigerant (b) during transitions between simultaneous and heating modes (S-mode and H-mode)



(a) (b)

Fig. 4. Measured temperatures of sources (a) and refrigerant (b) during transitions between simultaneous and cooling modes (S-mode and C-mode)

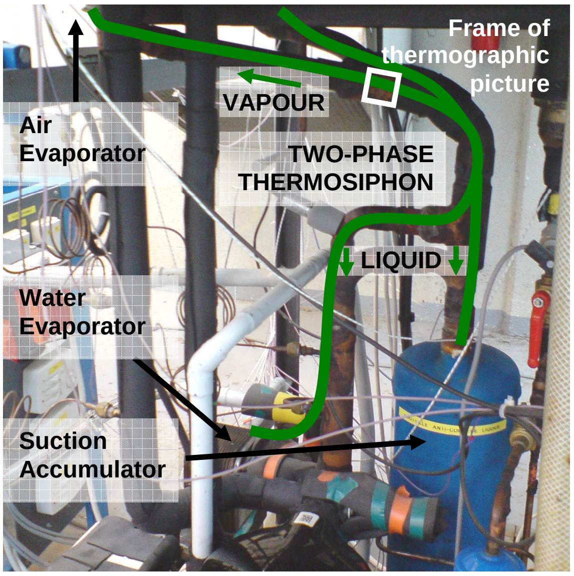
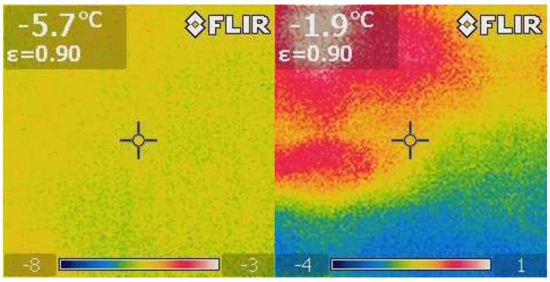


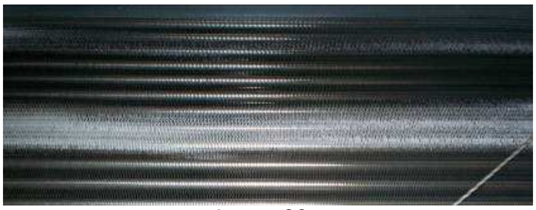
Fig. 5. Photograph of the thermosiphon setup



Before defrosting During defrosting

Fig. 6. Thermographic pictures of the tube between the two evaporators

time = 0 s



time = 60 s

time = 120 s

Fig. 7. Photographs of the air coil during the two-phase thermosiphon defrosting phase

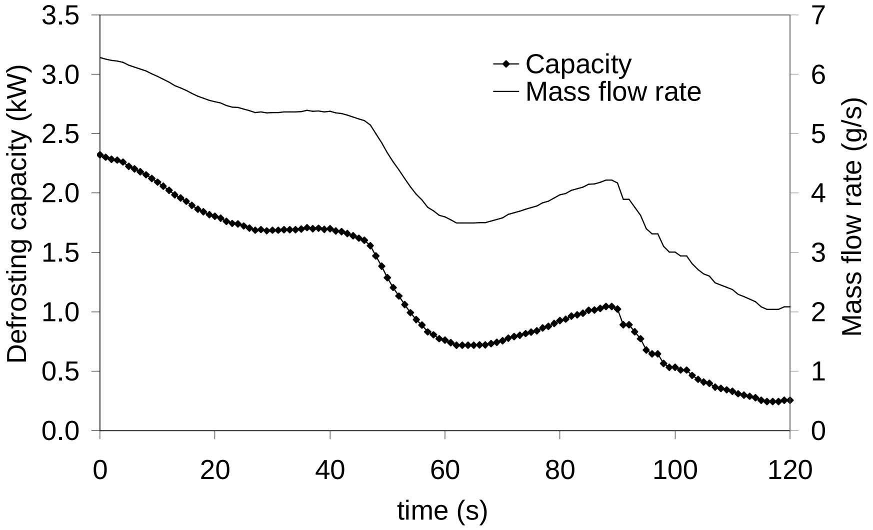


Fig. 8. Evolution of the thermosiphon defrosting capacity and refrigerant mass flow rate

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Designation	Value
Frost mass	397.6 g
Defrosting energy	133 kJ
Mean defrosting capacity	1.11 kW
Mean cooling capacity	15.31 kW
Mean heating capacity	17.72 kW
Mean refrigerant mass flow rate	5.30 $g.s^{-1}$

Table 1: Defrosting thermosiphon mean characteristics

Designation	**Value**
Condensing temperature (dew point)	33.5°C
Evaporating temperature (dew point)	11°C
Suction superheat	4°C
Nominal cooling capacity	19.00 kW
Nominal refrigerant mass flow rate	108 $g.s^{-1}$

Table 2: Operating conditions at the beginning of the defrosting sequence